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DESIGN CHARTS FOR CROSS-FLOW TUBULAR INTERCOOLERS

CHARGE-THROUGH-TUBE TYPE

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SUMMARY

On the basis of current heat-transfer theory, equations are developed relating the various dimensions, the air weight flow, and the performance of a cross-flow tubular intercooler in which the charge flows through and the cooling air across the tubes. These equations are then presented in graphical form in a series of design charts from which the intercooler design characteristics and performance can be quickly determined. A method of determining and presenting the performance of a given intercooler at various operating conditions is indicated.

Comparisons are made with the type of cross-flow tubular intercooler in which the cooling air flows through and the charge across the tubes. For a given charge and cooling-air pressure drop, air weight flow, number of tube banks, and tube weight, the two types generally have different values of cooling effectiveness and have different over-all dimensions. For operation at conditions that are fairly representative of aircraft intercooler practice, the charge-through-tube type usually requires more space than does the charge-across-tube type of intercooler.

INTRODUCTION

Cross-flow tubular intercoolers fall into two general classes: namely, (1) those in which the cooling air flows through and the charge across the tubes, and (2) those in which the cooling air flows across and the charge through the tubes. The selection of either of these types and its design varies with individual installations; the relative importance of intercooler size, weight, pressure drops, and ease of construction being the governing factor. The many variables involved in the design and performance of cross-flow tubular intercoolers tend to make the choice of the optimum type and its design very difficult. It is possible, however, to correlate these variables on the basis of heat-transfer theory in such a manner as to reduce this difficulty considerably. Design charts based on these correlations are presented in reference 1 for the type of cross-flow tubular intercooler in which the charge flows across tubes, the centers of which lie on the apexes of equilateral triangles (class 1). Since installation requirements, in some cases, would be better satisfied by an intercooler of the charge-through-tube type (class 2), a second set of similar charts has been prepared for this type with similarly arranged tubes and is presented in this report. An investigation based on published test data was made of the effect of other staggered tube arrangements on intercooler performance and size; the application of the design charts to these arrangements is discussed.

This work was conducted at the Langley Memorial Aeronautical Laboratory at Langley Field, Va., from August 1940 to January 1941.

SYMBOLS

H heat-transfer rate, Btu per second

h surface heat-transfer coefficient of air, Btu per second per square foot per °F

M rate of air flow, pounds per second

x distance along a tube, feet

l length in direction of air flow, feet

d_i inside tube diameter, feet

d_o outside tube diameter, feet

d average tube diameter, feet

$$\frac{1}{2}(d_i + d_o)$$

m number of tube banks in direction of flow across tubes

s minimum distance between walls of adjacent tubes measured perpendicular to direction of flow across tube banks, feet

s_p tube pitch measured parallel to direction of flow across tube banks, feet

V_{max} velocity through the minimum space s, feet per second

N number of tubes

f ratio of total cross-sectional area of intercooler tubes to area of charge core face

w width of intercooler block, feet

v volume of intercooler block, cubic feet

ϵ contractional-loss coefficient

g acceleration of gravity, feet per second per second

c_p specific heat of air at constant pressure (0.24 Btu per lb per $^{\circ}F$)

Δp total pressure drop across the intercooler, inches of water

ρ_0 standard atmospheric density (0.0765 lb per cu ft)

ρ air density, pounds per cubic foot

σ density of air relative to standard atmosphere (ρ/ρ_0)

P power required to force air across intercooler, horsepower

T_0 cooling-air temperature at intercooler entrance, $^{\circ}F$

$T_{1_{ex}}$ mean cooling-air temperature differential above T_0 at intercooler exit, $^{\circ}F$

T_2 charge temperature differential above T_0 at intercooler entrance, $^{\circ}F$

$T_{2_{ex}}$ mean charge temperature differential above T_0 at intercooler exit, °F

η cooling effectiveness, $\dfrac{T_2 - T_{2_{ex}}}{T_2}$

f''' friction factor

μ_f absolute viscosity of air film, pounds per foot per second

k_f thermal conductivity of air film, Btu per second per square foot per °F gradient per foot

β_1 ratio of mean cooling-air temperature rise to absolute cooling-air temperature at intercooler entrance $\left(\dfrac{T_{1_{ex}}}{T_0 + 460}\right)$

β_2 ratio of mean charge temperature drop to absolute charge temperature at intercooler entrance $\left(\dfrac{T_2 - T_{2_{ex}}}{T_2 + T_0 + 460}\right)$

Subscript 1 refers to cooling air

Subscript 2 refers to charge

Subscript en refers to entrance condition in intercooler

Subscript av refers to average conditions in intercooler

ANALYSIS

Cooling Effectiveness

With reference to figure 1, at some distance x along the length of an intercooler tube in the nth bank, the amount of heat per second dH exchanged between the charge and the cooling air through an elemental length of tube dx is

$$dH = h_2 \pi d_i \, dx \left(T_{2_x}{}^{(n)} - T_w\right) = h_1 \pi d_o \, dx \left(T_w - T_{1_{av}}{}^{(n)}\right)$$
$$= - \frac{M_2}{N} c_p \, dT_{2_x}{}^{(n)} \qquad (1)$$

where $T_{2_x}{}^{(n)}$ is the temperature differential above T_o of the charge at distance x along a tube in the nth bank and T_w is the temperature differential above T_o of the tube wall. Equation (1) involves the assumption that the local cooling air temperature at point x can be replaced by $T_{1_{av}}{}^{(n)}$ which is defined as the mean temperature differential above T_o of the cooling air across and over the length of a tube in the nth bank. The justification of this assumption will be discussed in detail later.

When T_w is eliminated in equation (1)

$$\frac{dT_{2_x}{}^{(n)}}{\left(T_{2_x}{}^{(n)} - T_{1_{av}}{}^{(n)}\right)} = - \frac{\pi h_1 d_o h_2 d_i N \, dx}{M_2 c_p \left(h_1 d_o + h_2 d_i\right)} \qquad (2)$$

Let

$$\frac{\pi h_1 d_o h_2 d_i N}{M_2 c_p \left(h_1 d_o + h_2 d_i\right)} = c' \qquad (3)$$

Then when equation (2) is integrated

$$T_{2_x}{}^{(n)} = T_2 + \left(T_{1_{av}}{}^{(n)} - T_2\right)\left(1 - e^{-c'x}\right) \qquad (4)$$

Also, the total heat given up by the charge to the nth bank of tubes is

$$H^{(n)} = \frac{M_2}{m} c_p \left(T_2 - T_2^{(n)}\right) = M_1 c_p \left(T_1^{(n)} - T_1^{(n-1)}\right) \quad (5)$$

where $T_2^{(n)}$ is the temperature differential above T_0 of the charge at the exit of a tube in the nth bank and $T_1^{(n)}$ and $T_1^{(n-1)}$ are the temperature differentials above T_0 of the cooling air after the nth and (n-1)th bank, respectively. The solution of equation (5) for $T_1^{(n)}$ is

$$T_1^{(n)} = T_1^{(n-1)} + r \left(T_2 - T_2^{(n)}\right) \quad (6)$$

where $r = \dfrac{M_2}{M_1 m}$

Then the average temperature differential above T_0 of the cooling air across the nth bank is

$$T_{1_{av}}^{(n)} = T_1^{(n-1)} + \frac{r}{2} \left(T_2 - T_2^{(n)}\right) \quad (7)$$

When equation (7) is substituted in equation (4) and the total length of the tube is considered,

$$T_2^{(n)} = T_2 + \left[T_1^{(n-1)} + \frac{r}{2} \left(T_2 - T_2^{(n)}\right) - T_2\right] \left(1 - e^{-c' l_2}\right) \quad (8)$$

In order to eliminate from equation (8) all terms $T_1^{(j)}$ where j ranges from o to $n - 1$, the following procedure is used:

From equation (6)

$$T_1^{(n-1)} = T_1^{(n-2)} + r \left(T_2 - T_2^{(n-1)}\right)$$

If in equation (8) this substitution is made and if the procedure is repeated $n - 1$ times, $T_2^{(n)}$ will be found to be a function of $T_2^{(j)}$ only. The result is written

$$T_2^{(n)} = T_2 \; (1 - n \, r \, \gamma + \gamma) + r \, \gamma \sum_{j=0}^{j=n-1} T_2^{(j)} \qquad (9)$$

where $\gamma = \dfrac{-\left(1 - e^{-c' l_2}\right)}{1 + \frac{r}{2}\left(1 - e^{-c' l_2}\right)}$

and $T_2^{(o)} = T_2$

For the first bank,

$$T_2^{(1)} = T_2(1 - r \, \gamma + \gamma) + r \, \gamma \sum_{j=0}^{j=0} T_2^{(j)}$$

$$= T_2(1 - r \, \gamma + \gamma) + r \, \gamma \, T_2$$

$$= (1 + \gamma) \; T_2$$

Likewise for the second bank,

$$T_2^{(2)} = T_2 \;\; (1 - 2 \, r \, \gamma + \gamma) + r \, \gamma \sum_{j=0}^{j=1} T_2^{(j)}$$

$$= T_2 \; (1 - 2 \, r \, \gamma + \gamma) + r \, \gamma \; \left[T_2 + T_2 \; (1 + \gamma)\right]$$

$$= (1 + \gamma + r \, \gamma^2) \; T_2$$

Also for the third bank,

$$T_2^{(3)} = T_2 \; (1 - 3r\gamma + \gamma) + r\gamma \left[T_2 + T_2 \; (1 + \gamma) + T_2 \; (1 + \gamma + r\gamma^2)\right]$$

$$= (1 + \gamma + 2r\gamma^2 + r^2\gamma^3) \; T_2$$

The foregoing operations indicate that for the mth bank

$$T_a^{(m)} = \left[1 + \gamma\ (1 + r\gamma)^{m-1}\right] T_a \tag{10}$$

The average exit charge temperature $T_{a_{ex}}$ is obtained upon proper summation

$$T_{a_{ex}} = \frac{\sum_{j=1}^{j=m} T_a^{(j)}}{m} = T_a + \frac{\gamma}{m}\left[\frac{1 - (1 + r\gamma)^m}{1 - (1 + r\gamma)}\right] T_a \tag{11}$$

on the assumption that the same quantity of air flows through each tube. The cooling effectiveness is

$$\eta = \frac{T_a - T_{a_{ex}}}{T_a} = \frac{1}{mr}\left[1 - (1 + r\gamma)^m\right]$$

or

$$\eta = \frac{M_1}{M_a}\left\{1 - \left[\frac{2}{1 + \frac{M_a}{2mM_1}\left(1 - e^{-c' l_a}\right)} - 1\right]^m\right\} \tag{12}$$

The exponent $c' l_a$ in equation (12) is evaluated in terms of the intercooler dimensions and the air weight flow by the same procedure as that followed in reference 1 for the equilateral tube arrangement. It will be shown later that within the range of available test data the surface heat-transfer coefficient h_1 and thus the exponent $c' l_a$ is only slightly changed by the use of other staggered tube arrangements. As in reference 1 the thermal conductivity and the absolute viscosity of the cooling air and the charge, as used in determining the heat-transfer coefficients, were evaluated at a constant value of 59° and 100° F, respectively.

$$c'_{l_2} = \frac{1.031 \times 10^{-2} \left(\frac{Nd}{M_1}\right)^{0.2} \left(\frac{M_1}{M_2}\right)^{0.2} \left(\frac{l_2}{d}\right)}{1 + 0.646 \left(\frac{M_2}{M_1}\right)^{0.8} \left(\frac{M_1}{Nd}\right)^{0.11} \left(\frac{s}{md}\right)^{0.69} \left(\frac{l_2}{d}\right)^{0.69}} \tag{13}$$

Expansion of the right-hand member of equation (12) shows that the variation in m after five banks, as it appears explicitly in that equation, has a negligible effect on the cooling effectiveness. The influence of m on the cooling effectiveness is through the exponent c'_{l_2}. The cooling effectiveness can then be stated as a function of M_1/M_2, l_2/d, md/s, and Nd/M_1, the same factors involved in the cooling effectiveness expression given in reference 1 for the type of cross-flow tubular intercooler in which the charge flows across the tubes and the cooling air, through the tubes.

In order to make graphical representation less complicated, the effect of Nd/M_1 may be accounted for by letting

$$\left(\frac{l_2}{d}\right)\left(\frac{Nd}{M_1}\right)^{0.2} = 10^{0.2}\left(\frac{l_2}{d}\right)_{eq}$$

and

$$\left(\frac{md}{s}\right)\left(\frac{Nd}{M_1}\right)^{0.36} = 10^{0.36}\left(\frac{md}{s}\right)_{eq}$$

where the subscript eq means equivalent. Equation (13) can now be written

$$c'_{l_2} = \frac{1.634 \times 10^{-2} \left(\frac{M_1}{M_2}\right)^{0.2} \left(\frac{l_2}{d}\right)_{eq}}{1 + 0.502 \left(\frac{M_2}{M_1}\right)^{0.8} \left(\frac{s}{md}\right)_{eq}^{0.69} \left(\frac{l_2}{d}\right)_{eq}^{0.69}} \tag{14}$$

Figures based on equations (12) and (14) will be presented to show the effects of the factors M_1/M_2, $(l_2/d)_{eq}$, and

$(md/s)_{eq}$ on the cooling effectiveness. The corrections to be applied to l_2/d and md/s to obtain $(l_2/d)_{eq}$ and $(md/s)_{eq}$ will be presented in graphical form as functions of Nd/M_1.

Pressure Drop of Cooling Air

The pressure drop due to friction in air flowing across staggered banks of tubes the centers of which lie on the apexes of equilateral triangles is evaluated by the same method used in reference 1. It will be shown later that within the range of available test data the friction factor f''' and thus the cooling-air pressure drop Δp_1 is changed only slightly by the use of other staggered tube arrangements. As in reference 1 the viscosity of the cooling air is evaluated at 59° F.

$$\frac{\rho_{1_{av}}}{\rho_0} \Delta p_1 = \frac{0.1324 m^{0.78}}{5.2\ \rho_0 g} \left(\frac{M_1}{N d\, l_2}\right)^{1.78} \left(\frac{md}{s}\right)^2 \left(\frac{l}{d}\right)^{0.22} \qquad (15)$$

Pressure Drop of Charge

By the procedure followed in reference 1 to determine the pressure drop of air through the intercooler tubes, the following equation is obtained upon addition of the entrance, fluid-frictional, heating, and exit pressure changes:

$$\frac{\rho_{2_{av}}}{\rho_0} \Delta p_2 = 0.0633A\left(\frac{M_2}{N d\, l_2}\right)^2 \left(\frac{l_2}{d}\right)^2$$

$$+ 0.00124 \left(\frac{M_2}{N d\, l_2}\right)^{1.8} \left(\frac{l_2}{d}\right)^{2.8} \left(\frac{l}{d}\right)^{0.2} \qquad (16)$$

where

$$A = \left(\frac{2 - \beta_2}{2 - 2\beta_2}\right)(\epsilon - 1.09f^2 - 0.91) + (2 - \beta_2)\ (f^2 - f + 1)$$

and

$$\beta_2 = \frac{T_2 - T_{2\,ex}}{T_2 + T_0 + 460} = \frac{\eta}{\left(1 + \frac{T_0 + 460}{T_2}\right)}$$

As in reference 1 the viscosity of the charge is evaluated at 100° F. A plot of f is given in reference 1 for the equilateral tube arrangement. The factor ϵ is a function only of f and is plotted in reference 1. The factor β_2 is introduced to evaluate the heating pressure regain and to give the entrance and exit pressure changes in terms of the mean charge density. A change in staggered tube arrangement from the equilateral spacing causes a change in f and thus a change in the entrance-exit pressure drop. This pressure drop, however, is usually a small portion of the total drop, especially for long tubes.

Cooling-Air Power Loss

By the procedure followed in reference 1, the expression for the power required to force cooling air across the intercooler is,

$$\frac{\sigma_{1_{av}}{}^{2}P_1}{M_2} \quad \frac{M_1}{M_2}\,\frac{5.2\sigma_{1_{av}}\Delta p_1}{550\rho_0} \tag{17}$$

When the power is expressed as a percentage of the engine brake horsepower for a fuel-air ratio of 0.08 and a specific brake fuel consumption of 0.5 pound per brake horsepower per hour,

$$\frac{\sigma_{1_{av}}{}^{2}P_1\ (100)}{\text{brake horsepower}} \quad \frac{100}{576}\left(\frac{\sigma_{1_{av}}{}^{2}P_1}{M_2}\right)$$

Charge Power Loss

The power required to force the charge through the intercooler is given in reference 1 as

$$\frac{\sigma_{2_{av}}\,{}^{a}p_2}{M_2} = \frac{5.2\sigma_{2_{av}}\,\Delta p_2}{550\ \rho_0} \tag{18}$$

and for a fuel-air ratio of 0.08 and a specific brake fuel consumption of 0.50 pound per brake horsepower per hour

$$\frac{\sigma_{2_{av}}\,{}^{a}p_2\,.(100)}{\text{brake horsepower}} \quad \frac{100}{576}\left(\frac{\sigma_{2_{av}}\,{}^{a}p_2}{M_2}\right).$$

The relative densities (σ) in the foregoing equations are average densities. The relation between the average and the entrance densities, when the effects of pressure changes in the intercooler are neglected, may be expressed as follows:

$$\sigma_{1_{av}} = \sigma_{1_{en}}\left(\frac{2 + \beta_1}{2 + 2\beta_1}\right) \tag{19}$$

where $\beta_1 = \dfrac{(M_2/M_1)\ \eta T_2}{T_0 + 460}$

$$\sigma_{2_{av}} = \sigma_{2_{en}}\left(\frac{2 - \beta_2}{2 - 2\beta_2}\right) \tag{20}$$

where $\beta_2 = \dfrac{\eta T_2}{T_2 + T_0 + 460}$

Intercooler Dimensions

The dimensions of this type of intercooler may be expressed as follows:

The dimension of the block in the direction of the cooling air flow is

$$l_1 = s_p\ (m - 1) + d_o$$

where, for the equilateral arrangement, $s_p = 0.866\ (s + d_o)$

The width of the block is

$$w = \left(\frac{N}{m} + \frac{1}{2}\right)\ (s + d_o)$$

The volume of the block is therefore

$$V = l_1\ l_2\ w$$

where l_2 is the dimension in the direction of the charge flow (tube length).

DISCUSSION

The Cooling Effectiveness Equation

In the integration of equation (2) the local cooling-air temperature differential above T_o at a point along a tube in the nth bank was replaced by $T_{1_{av}}(n)$, the average temperature differential above T_o of the cooling air across and over the length of a tube in the nth bank. This assumption is valid only if the air along a given tube completely mixes, thus equalizing the temperature of the cooling air along that tube. Since some mixing probably occurs, the true condition lies between that assumed in this report and the condition assumed by Nusselt (reference 2) that no temperature equalization occurs along a tube. If, however, equation (12) is compared with Nusselt's expression for the cooling effectiveness (see reference 2), it is found that: (a) equation (12) is relatively simple; and (b) in the usual range of effectiveness values used in intercoolers, the results obtained from equation (12) agree fairly well with Nusselt's results.

A comparison of the effectiveness values derived by the two expressions is given in figure 2. This figure shows that over a wide range of effectiveness values covered by the design charts presented in this report, the two analyses give the same results. In the range of higher values of cooling effectiveness, however, the Nusselt analysis gives slightly higher values, the maximum difference in the range of the design charts being 4.0 percent effectiveness. The values given by equation (12) are used in preparing the design charts; they make for slightly conservative values of cooling effectiveness which may be corrected to the Nusselt values, if desired, by the use of figure 2.

Effects of Primary Intercooler Variables

Figure 3, which is a plot of equations (12) and (14), shows that the cooling effectiveness increases with M_1/M_2, $(l_2/d)_{eq}$, and $(md/s)_{eq}$ at a rate that diminishes as these variables increase. The effect of l_2/d and md/s on the cooling effectiveness is nearly the same as that of $(l_2/d)_{eq}$ and $(md/s)_{eq}$. The corrections relating the actual l_2/d and md/s to $(l_2/d)_{eq}$ and $(md/s)_{eq}$, respectively, are replotted against Nd/M_1 in figure 4 from reference 1.

Equations (15) and (16) indicate that an increase in either l_2/d, md/s, or M_1/M_2, for a given M_2 in an attempt to attain higher values of cooling effectiveness, is accompanied by an increase in pressure drop through the intercooler or across it. In the selection of an intercooler these pressure drops and the consequent power expenditures as well as the intercooler weight and size must be considered.

Design Charts

Although equations (12), (13), (15), and (16) may be used to predict intercooler performance when the intercooler dimensions and weight flows of the charge and the cooling air are known, their application would prove quite cumbersome. By the introduction of simplifications identical with those of reference 1, these equations are represented graphically in the form of charts (figs. 5 to 10) readily usable by the designer. These simplifications detract little from the validity of the charts since the resulting errors are well within the limit of experimental accuracy.

The basic variables used in the construction of the charts are l_2/d, md/s, M_1/M_2, Ndl_2/M_2, m, $\sigma_{1_{av}}\Delta p_1$, and $\sigma_{2_{av}}\Delta p_2$. The main design charts (figs. 5, 6, and 7) give for various values of M_1/M_2 and l_2/d the cooling effectiveness plotted against Ndl_2/M_2, which is an index of heat-transfer surface and thus of tube weight. Each figure is given for a constant value of m (5, 20, and 30 banks of tubes, respectively) and covers the range of $\sigma_{1_{av}}\Delta p_1$ from 2 inches to 8 inches of water. Linear interpolation for intermediate values of m and $\sigma_{1_{av}}\Delta p_1$ gives results well within experimental accuracy. The main design charts are calculated for a constant value of $\sigma_{2_{av}}\Delta p_2$ (5.5 inches of water). Because variation in β_2 has a negligible effect on the values in these charts a constant value for β_2 of 0.20, which represents a usual condition, was used in the preparation of the charts.

With the aid of figures 8 and 9(a) these design charts may be used for other values of $\sigma_{2_{av}}\Delta p_2$. Figure 8 gives the correction to be added algebraically to the chart values of l_2/d for various values of $\sigma_{2_{av}}\Delta p_2$ and Ndl_2/M_2. Likewise figure 9(a) contains the corrections to be added algebraically to the chart values of cooling effectiveness η.

The tables included in the main design charts give the power expenditures involved in the operation of an intercooler. Figure 9(b) gives the variation in the power required to force the charge through the intercooler with charge pressure drop $\sigma_{2_{av}}\Delta p_2$. Aside from these power expenditures are those due to the intercooler weight, duct losses, and reduction in manifold pressure - all of which depend on installation and flight conditions and should be considered in the choice of the optimum intercooler.

Some energy may be recovered from the cooling-air stream by means of the Meredith effect.

Figure 10(a) gives, for an intercooler with 5 banks of tubes, a plot of md/s against $\frac{Nd l_2}{M_2} \Big/ \frac{M_1}{M_2}$ for various values of $\sigma_{1_{av}} \Delta p_1$. For given values of $\frac{Nd l_2}{M_2} \Big/ \frac{M_1}{M_2}$ and $\sigma_{1_{av}} \Delta p_1$ a change in the number of tube banks of an intercooler must be accompanied by a change in md/s. This change is given in figure 10(b) in the form of a plot of $\left(\frac{md}{s}\right)_m \Big/ \left(\frac{md}{s}\right)_5$ against m where the subscript on md/s denotes the number of tube banks for which that value of md/s applies.

Effect of Tube Arrangement on Intercooler Characteristics

Figures 11 and 12 are plots of data obtained from references 3, 4, and 5, and show the effect of staggered tube arrangement on the surface heat-transfer coefficient h_1 and the friction factor f'''. In these figures $h_1 d_o/k_{1_f}$ and f''' are plotted against the ratio $s_p/(s + d_o)$ for various values of s/d_o and Reynolds number. The Reynolds number in figure 11 is based on the tube diameter, and in figure 12 it is based on the dimension s. In each of these figures the equilateral tube arrangement is indicated by the vertical line at $s_p/(s + d_o) = 0.866$ on which are marked the values of $h_1 d_o/k_{1_f}$ and f''' obtained from reference 5 and used in this report to evaluate intercooler performance.

It is shown in figures 11 and 12 that $h_1 d_o/k_{1_f}$ and f''' are nearly constant for a large range of values of $s_p/(s + d_o)$ and are approximately equal to the values of

$h_1 d_o/k_{1_f}$ and f''' given by reference 5 for equilateral spacing. Thus the design charts, although based on the data of reference 5, apply with a good degree of accuracy for a large range of values of $s_p/(s + d_o)$. It is evident that the intercooler dimension in the direction of cooling-air flow l_1, and hence the intercooler volume v, can be decreased by decreasing $s_p/(s + d_o)$ with little change in intercooler performance. There is probably a lower limit to $s_p/(s + d_o)$ beyond which further reduction in this dimension results in an undesirable change in intercooler performance. The range of the available data is not sufficiently large to show this lower limit except possibly in the case of $s/d_o = 2$, where the Nusselt number appears to decrease markedly below the value of $s_p/(s + d_o) = 0.4$. The smallest values of $s_p/(s + d_o)$ and the corresponding values of s/d_o covered by the test data of references 3 and 4 and shown in figures 11 and 12 are given in the following table:

s/d_o	0.25	0.5	1.0	2.0
$s_p/(s + d_o)$	1.00	.67	.45	.20

In the case of $s/d_o = 0.25$ the value of $s_p/(s + d_o)$ could have been reduced from 1.00 to 0.866 (equilateral spacing) with little change in $h_1 d_o/k_{1_f}$ or f'''. In reference 1 tests are reported in which the value of s/d_o was 0.038 for the equilateral tube arrangement, and the results agreed with the equations for $h_1 d_o/k_{1_f}$ and f''' presented in reference 5.

Attention is called to the fact that the surface heat-transfer coefficient shown in figure 11 represents the surface heat transfer on the outside of the tubes only, and that variations in the cooling effectiveness of an intercooler are not as great as the variations noted in this heat-transfer coefficient. Figure 13 shows the ratio of percentage change α $(= d\eta/\eta)$ in cooling effectiveness to the percentage change ϕ $(= dh_1/h_1)$ in the outer-surface heat-transfer coefficient. These curves are based on equation (12). According to this figure, in the usual range of intercooler practice the percentage change in η is less than that in h_1 by a factor of 0.5 or less.

It appears that considerable latitude in s_p is permitted for a given value of s without serious change in intercooler performance but with very appreciable variation in intercooler volume. Consequently the design charts presented in this report are not limited to equilateral tube arrangement but should also be applicable with a fair degree of accuracy to other tube arrangements within the extent of the evidence presented.

Although hidden by the dispersion of the data (fig. 11) there are probably optimum values for s_p which provide slightly better performance than the equilateral spacing. It is believed that further tests on the effect of s_p on heat transfer and pressure drop covering a larger range than the data given in references 3 and 4 should be made in order to determine (a) the values of s_p for maximum performance and (b) the values of s_p below which the performance decreases rapidly.

Illustrations of the Use of the Intercooler Design Charts

Case I

Let it be supposed that an intercooler is to be designed for the following set of conditions:

Engine and supercharger characteristics:

(1) Charge mass flow M_2, pounds per second - - - - - - - - - - - - - 1.75

(2) Charge temperature at supercharger outlet, $^{\circ}F$ - - - - - - - - - - - - 160

(3) Charge pressure at supercharger outlet, inches of mercury absolute - - - - - - - - - - - - - - 40

Desired intercooler dimensions:

(4) Average tube diameter d, feet - - - $\frac{1}{48}$

(5) Number of tube banks m - - - - - - 20

(6) Tube-wall thickness t, feet (copper density ρ_t, 555 lb/cu ft) - - - - - - - - - - - - 0.0005

Intercooler limitations:

(7) Cooling-air pressure drop Δp_1, inches of water - - - - - - - - - 8.0

(8) Charge pressure drop Δp_2, inches of water - - - - - - - - - - - - - - 4.5

Desired intercooler performance at 21,000 feet altitude:

(9) Cooling effectiveness η, percent - 60

The intercooler design will be made for $M_1/M_2 = 2$.

It is desired to find the following intercooler characteristics:

Tube length l_2

Number of tubes N

Transverse tube spacing s

Parallel tube spacing s_p

Weight of intercooler tubes W_t and dimensions of intercooler block

Power required to force cooling air across the intercooler tube banks P_1

Power required to force charge through the intercooler P_2

(10) From a table of standard altitude at 21,000 feet

$$\sigma_{1_{en}} = 0.52$$

$$T_o = -16^\circ \text{ F}$$

(11) From items (2), (9), and (10) and from equation (19) for $M_1/M_2 = 2$

$$\beta_1 = \frac{\frac{1}{2} \times 0.6 \times 176}{460 - 16} = 0.119$$

so that

$$\frac{\sigma_{1_{av}}}{\sigma_{1_{en}}} = \frac{2.119}{2.238} = 0.95$$

(12) From items (7), (10), and (11)

$$\sigma_{1_{av}} \Delta p_1 = 0.95 \times 0.52 \times 8 = 4 \text{ inches of water}$$

(13) From items (2) and (3)

$$\sigma_{2_{en}} = \frac{519}{620} \times \frac{40}{30} = 1.115$$

(14) From items (2), (9), and (10)

$$\beta_2 = \frac{0.6 \times 176}{176 + 460 - 16} = 0.17$$

so that

$$\frac{\sigma_{2_{av}}}{\sigma_{2_{en}}} = \frac{1.83}{1.66} = 1.10$$

(15) From items (8), (13), and (14)

$$\sigma_{2_{av}} \Delta p_2 = 1.10 \times 1.115 \times 4.5 = 5.5 \text{ inches of water}$$

(16) If figure 6(b) which applies for $m = 20$,

$\sigma_{1_{av}} \Delta p_1 = 4$, and $\sigma_{2_{av}} \Delta p_2 = 5.5$ is used,
for $M_1/M_2 = 2$ and $\eta = 60$

$$l_2/d = 127$$

and

$$Ndl_2/M_2 = 28.9$$

(17) From items (4) and (16)

$$l_2 = 2.65 \text{ feet}$$

(18) From items (1), (4), (16), and (17)

$$N = \frac{28.9 \times 1.75}{2.65 \times \frac{1}{48}} = 916 \text{ tubes}$$

(19) If figure 10(a) and items (12) and (16) are used for $M_1/M_2 = 2$

$$(md/s)_5 = 75$$

(20) If figure 10(b) and items (5) and (19) are used

$$(md/s)_{20} \Big/ (md/s)_5 = 0.582$$

so that

$$(md/s)_{20} = 43.5$$

(21) From items (4), (5), and (20)

$$s = \frac{20 \times \frac{1}{48}}{43.5} = 0.00958 \text{ foot}$$

(22) From items (4), (6), and (21)

$s/d_o = \frac{s}{d + t} = 0.45$, so that, from figure 11,

within the range given, the lowest value of

$s_p/s + d_o = 0.67$ and thus

$s_p = 0.67\ (0.00958 + 0.0213) = 0.0207$ foot

(22a) For the equilateral arrangement

$s_p = 0.866\ (0.00958 + 0.0213) = 0.0268$ foot

(23) From items (1), (6), and (16) the weight of the intercooler tubes is

$(0.0005)\ (28.9)\ (1.75)\ (555\pi) = 44.2$ pounds

(24) Dimension of intercooler block in direction of cooling-air flow from items (4), (5), (6), and (22) is

when $\frac{s_p}{s + d_o} = 0.67$

$(0.0207 \times 19) + 0.0213 = 0.41$ foot

when $\frac{s_p}{s + d_o} = 0.866$ (equilateral arrangement)

$(0.0268 \times 19) + 0.0213 = 0.53$ foot

(25) Width of intercooler block from items (4), (5), (6), (18), and (21) is

$\left(\frac{916}{20} + \frac{1}{2}\right)\ (0.0213 + 0.00958) = 1.430$ feet

(26) Volume of intercooler block when

$\frac{s_p}{s + d_o} = 0.67$ from items (17) (24), and (25) is

$$2.65 \times 0.41 \times 1.415 = 1.537 \text{ cubic feet}$$

and when $\frac{s_p}{s + d_o} = 0.866$ is

$$2.65 \times 0.53 \times 1.415 = 1.987 \text{ cubic feet}$$

(27) From table at top of figure 6(b) for $M_1/M_2 = 2$

$\sigma_{1_{av}}{}^2 P_1/M_2 = 0.99$ horsepower per pound per second of charge flow

(28) Also,

$\sigma_{2_{av}}{}^2 P_2/M_2 = 0.68$ horsepower per pound per second of charge flow

(29) From items (1), (10), (11), and (27)

$$P_1 = \frac{0.99 \times 1.75}{(0.95 \times 0.52)^2} = 7.10 \text{ horsepower}$$

(30) From items (1), (13), (14), and (28)

$$P_2 = \frac{0.68 \times 1.75}{(1.115 \times 1.10)^2} = 0.79 \text{ horsepower}$$

In figure 2 it is shown that the value of η from the Nusselt analysis for $M_1/M_2 = 2$ is 1 percent higher than the value 60 used in this example.

Case II

The effect on intercooler dimensions and performance of a change in charge pressure drop $\sigma_{2_{av}}\Delta p_2$ from 5.5

inches of water, for which the main design charts have been drawn, to another value is illustrated by the following example:

In case I suppose that $\sigma_{2_{av}} \Delta p_2$ is increased to 8.0 inches of water; the engine and supercharger characteristics, the desired intercooler dimensions, the remaining intercooler limitation, the altitude, and the value of M_1/M_2 being kept the same.

(31) If figure 8 and item (16) are used

$$\Delta \frac{l_2}{d} = 19$$

(32) From figure 9(a) and for $M_1/M_2 = 2$

$$\Delta\eta = 2 \text{ percent}$$

so that

$$\eta = 60 + 2 = 62 \text{ percent}$$

(33) From items (16) and (31)

$$\frac{l_2}{d} = 127 + 19 = 146$$

and

$$l_2 = 3.05 \text{ feet}$$

(34) From items (1), (4), (16), and (33)

$$N = \frac{23.9 \times 1.75}{3.05 \times \frac{1}{48}} = 796 \text{ tubes}$$

(35) Inasmuch as $Nd l_2/M_2$ is kept constant with $\sigma_{2_{av}} \Delta p_2$, the weight of the intercooler tubes remains the same.

(36) The tube spacing does not change because Ndl_2/M_2, M_1/M_2, $\sigma_{1_{av}}\Delta p_1$, and m are the same. Thus the dimension of the intercooler block in the direction of cooling-air flow is the same; the width changes inversely with the change in tube length, thereby keeping the intercooler volume the same.

(37) If figure 9(b) is used

$$\sigma_{2_{av}} \frac{2\,P_2}{M_2} = 1.00 \text{ horsepower per pound per}$$

second of charge flow

(38) From items (12), (24), and (37)

$$P_2 = \frac{1.00 \times 1.75}{(1.115 \times 1.10)^2} = 1.16 \text{ horsepower}$$

(39) The cooling-air weight flow and the attendant power loss across the intercooler remain the same as in case I.

Performance Charts

It has been shown how an intercooler is designed and how the performance is predicted for a given set of operating conditions. Although the design charts can also be used to predict the performance of the designed intercooler at other operating conditions - for example, operation at another altitude or perhaps operation with a different weight of charge flowing through the tubes, there is a simpler and more direct method involving the use of figures 3 and 4 and the relations between pressure drop and weight flow of charge and cooling air. The information can then be presented in convenient form for use by designers to determine the intercooler characteristics at any operating condition. The procedure for obtaining an intercooler performance chart is summarized in the following outline and an illustrative example presented.

1. Relation between $\sigma_{1_{av}}\Delta p_1$ and M_1

   Equation (15) shows that $\sigma_{1_{av}}\Delta p_1$ varies directly with $M_1^{1.8}$.

2. Relation between $\sigma_{2_{av}}\Delta p_2$ and M_2

   The exact relation between $\sigma_{2_{av}}\Delta p_2$ and M_2 (equation 16) is quite involved because a portion of the pressure drop (entrance, exit, and heating losses) varies with M_2^2 while another portion (tube frictional loss) varies with $M_2^{1.8}$. In symbolic form for a given intercooler,

$$\sigma_{2_{av}}\Delta p_2 = M_2^{1.8}\left(K_1 + K_2 M_2^{0.2}\right)$$

   where K_1 and K_2 are constants.

   The values of K_1 and K_2 can be found from the intercooler dimensions and thus the relation between $\sigma_{2_{av}}\Delta p_2$ and M_2 can be definitely established. Because the frictional loss in the tubes is a large portion of the total loss (in the intercooler of case I for $\sigma_{2_{av}}\Delta p_2 = 5.5$ inches of water the tube frictional loss was 87 percent of the total loss), the simpler yet sufficiently accurate expression,

$$\sigma_{2_{av}}\Delta p_2 = K_3 M_2^{1.8}$$

   where K_3 is a constant, may be used.

3. Relation of η to $\sigma_{1_{av}}\Delta p_1$ and $\sigma_{2_{av}}\Delta p_2$

   For a given intercooler $\sigma_{1_{av}}\Delta p_1$ determines M_1 and Nd/M_1, which, in turn, fix the values of

$(l_2/d)_{eq}$ and $(md/s)_{eq}$. (See fig. 4.) From figure 3, for the given $(l_2/d)_{eq}$ and $(md/s)_{eq}$, η can be plotted against M_1/M_2. Then for every cooling effectiveness value there is a corresponding value of M_2 and thus a corresponding value of $\sigma_{2_{av}}\Delta p_2$. This procedure is repeated for various values of $\sigma_{1_{av}}\Delta p_1$.

4. Procedure for drawing performance charts

The performance chart gives for a certain intercooler the relations of η to $\sigma_{1_{av}}\Delta p_1$ and $\sigma_{2_{av}}\Delta p_2$, of $\sigma_{1_{av}}\Delta p_1$ to M_1, and of $\sigma_{2_{av}}\Delta p_2$ to M_2. In order to increase the scope of the performance chart, the air weight flows are given per unit width of intercooler. Thus the given intercooler is assumed to consist of a number of intercoolers of unit width placed in parallel. Changing the width of the given intercooler by changing the number of tubes per bank is equivalent to changing the number of intercoolers of unit width placed in parallel. The performance chart, therefore, can be used for any value of w provided the change in w is due only to the change in the number of tubes per bank, the tube spacing and other dimensions being constant. Figure 14 is a convenient plot of the foregoing relations for the intercooler designed in case I. In this figure $\sigma_{1_{av}}\Delta p_1$ is plotted against $\sigma_{2_{av}}\Delta p_2$ for various values of η. Also M_2/w is plotted against $\sigma_{2_{av}}\Delta p_2$, and M_1/w against $\sigma_{1_{av}}\Delta p_1$.

The best method of illustrating the procedure for making a performance chart is to follow through a sample problem for the intercooler designed in case I.

The dimensions of the intercooler of case I are:

$d = 1/48$ foot	$s = 0.00958$ foot
$m = 20$ banks	$t = 0.0005$ foot
$l_2 = 2.65$ feet	$w = 1.415$ feet
$N = 916$ tubes	

When $\sigma_{1_{av}} \Delta p_1 = 4$ inches of water, $\sigma_{2_{av}} \Delta p_2 = 5.5$ inches of water and $M_2 = 1.75$ pounds per second, the intercooler has a cooling effectiveness of 60 percent and a value of $M_1/M_2 = 2$ so that $M_1 = 3.5$ pounds per second. The problem is to determine the performance of the intercooler at other values of $\sigma_{1_{av}} \Delta p_1$ and $\sigma_{2_{av}} \Delta p_2$ and to present this information in the convenient manner explained before. When $\sigma_{1_{av}} \Delta p_1 = 4$ inches of water, $M_1 = 3.5$ pounds per second, $M_1/w = 2.47$ pounds per second per foot width of intercooler, and $Nd/M_1 = 5.5$ feet per pound per second.

For $l_2/d = 127$ and $md/s = 43.5$ from figure 4, $(l_2/d)_{eq} = 113$ and $(md/s)_{eq} = 34.8$.

From figure 3 the effectiveness values corresponding to the various values of M_1/M_2 can be chosen.

M_1/M_2	η
0	0
1	44.0
2	60.2
4	75.5

From the curve obtained by plotting these points the value of M_1/M_2 can be selected for any value of cooling effectiveness for the given $(l_2/d)_{eq}$ and $(md/s)_{eq}$.

Thus for every value of cooling effectiveness there is a corresponding value of M_2 for the given value of M_1. From the fact that $\sigma_{2_{av}}\Delta p_2 = 5.5$ inches of water when $M_2 = 1.75$ pounds per second and from the relation

$\sigma_{2_{av}}\Delta p_2 = K_3 M_2^{1.8}$, a plot of M_2 versus $\sigma_{2_{av}}\Delta p_2$ can be drawn. Thus for any value of M_2 the corresponding value of $\sigma_{2_{av}}\Delta p_2$ can be found as follows:

η	M_1/M_2	M_2	$\sigma_{2_{av}}\Delta p_2$	M_2/w
50	1.27	2.75	12.4	1.94
55	1.57	2.23	8.4	1.58
60	1.96	1.79	5.7	1.27
65	2.45	1.43	3.8	1.01
70	3.12	1.12	2.5	.79
75	3.93	.89	1.7	.63

For any other value of $\sigma_{1_{av}}\Delta p_1$ a solution for M_1 can be obtained from the relation $\sigma_{1_{av}}\Delta p_1 = K_4 M_1^{1.8}$. The procedure is then repeated for various values of $\sigma_{1_{av}}\Delta p_1$ and M_1.

The following table gives the results of the forego-ing procedure. Figure 14 is a plot of the results in the recommended form.

	(a)	(b)	(c)
$\sigma_{1_{av}} \Delta p_1$, in. of water	2	6	8
M_1, lb/sec	2.38	4.38	5.13
M_1/w, lb/sec/ft	1.68	3.10	3.63
Kd/M_1, ft/lb/sec	8.0	4.4	3.7
$(l_2/d)_{eq}$	121	108	104
$(md/s)_{eq}$	40.0	32.6	30.9

M_1/M_2	η, percent		
0	0	0	0
1	46	43	42
2	63	59	58
4	78	74	73

η	M_1/M_2	K_2	$\sigma_{2_{av}} \Delta p_2$	M_2/w
(a)				
50	1.18	2.02	7.1	1.43
55	1.47	1.62	4.7	1.15
60	1.82	1.31	3.3	.926
65	2.25	1.06	2.2	.750
70	2.32	.84	1.5	.594
75	3.50	.68	1.0	.480
(b)				
50	1.36	3.22	16.4	2.28
55	1.68	2.61	11.1	1.85
60	2.11	2.08	7.5	1.47
65	2.63	1.67	5.0	1.18
70	3.35	1.31	3.3	.926
(c)				
50	1.42	3.61	20.2	2.55
55	1.77	2.90	13.6	2.05
60	2.20	2.33	9.1	1.65
65	2.77	1.85	6.0	1.31
70	3.50	1.47	4.0	1.04

Comparison of Charge-Through-Tube and Charge-Across-Tube Intercoolers

In reference 1 the cooling effectiveness of a cross-flow tubular intercooler of the charge-across-tube type is shown to be a function mainly of M_1/M_2 and cl_2. From the definitions of c and c',

$$cl_2 = \frac{c'l_2}{\frac{M_1}{M_2}}$$

where $c'l_2 = \frac{hS}{M_2c_p}$

and where h is the over-all heat-transfer coefficient and S is the heat-transfer area. The cooling effectiveness can then be stated simply as a function of M_1/M_2 and $c'l_2$, the same terms involved in Nusselt's analysis and in equation (12) of this report.

For a given M_1/M_2 and $c'l_2$ and within the range of the design charts presented in this report, the cooling effectiveness of the charge-across-tube type of cross-flow tubular intercooler agrees closely with the cooling effectiveness values obtained from Nusselt and from equation (12) of this report, the small differences being due to the difference in the assumptions made in the derivations. Thus for a given heat-transfer area, over-all heat-transfer coefficient, and weight flow of charge and cooling air, the values of effectiveness of the two types of cross-flow tubular intercoolers are the same and are equal to the effectiveness of heat transfer due to the cross flow of two fluids over a plate.

For a given cross-flow tubular intercooler (that is, a given m, d, s, l_2, and N) operating at $M_1/M_2 = 1$, the process of interchanging the flows (that is, first operating the intercooler as a charge-across-tube type and then as a charge-through-tube type or vice versa) does not change the cooling effectiveness if the effects of the viscosities and the thermal conductivities of the charge and the cooling air, which in reference 1 were shown to be small, are neglected. The pressure drop across the intercooler tubes and the pressure drop through the tubes remain substantially the same. Expressed in symbolic form,

$$\eta_t = \eta_a$$

$$\left(\sigma_{1_{av}} \Delta p_1\right)_t = \left(\sigma_{2_{av}} \Delta p_2\right)_a$$

and

$$\left(\sigma_{1_{av}} \Delta p_1\right)_a = \left(\sigma_{2_{av}} \Delta p_2\right)_t$$

where the subscripts a and t denote the charge-across-tube type and charge-through-tube type of intercooler, respectively. If M_1/M_2 has a value other than unity, however, the process of interchanging the flows increases one of the two individual heat-transfer coefficients and decreases the other, the net effect on the over-all heat-transfer coefficient and thus on the cooling effectiveness depending mainly on the initial values of the individual heat-transfer coefficients and the value of M_1/M_2. The pressure drops through the tubes and across the tubes change so as to satisfy approximately the relationship

$$\sigma_{av} \, \Delta p = K \, M^{1.8}$$

where K is a constant of proportionality and M either the weight flow of the charge or the weight flow of the cooling air.

For values of M_1/M_2 other than unity, in order that the two types of intercoolers have equal values of cooling effectiveness for a given heat-transfer area and given air flows, the disposition of the heat-transfer area and thus the over-all dimensions of the two intercoolers must be different since the flow conditions and individual heat-transfer coefficients are different. In general, the pressure drops of the charge and the cooling air must also be different. For $m = 5$ banks, $\sigma_{2_{av}} \Delta p_2 = 10$ inches of water, and $\sigma_{1_{av}} \Delta p_1 = 2$ inches and 8 inches of water, figure 15 gives the values of cooling effectiveness of both types of intercoolers plotted against $Nd l_2/M_2$ for various values of M_1/M_2. The effect of the number of tube banks on the relations between the cooling effectiveness values for both types of intercooler is small so that figure 15 covers conditions which are fairly representative of aircraft intercooler practice. The cooling effectiveness of the charge-across-tube type increases at a slightly faster rate with M_1/M_2 than does the cooling effectiveness of the other type. The rate of change in cooling effectiveness with $Nd l_2/M_2$ is approximately the same for both types of intercoolers. From figure 15 it is seen that some value of M_1/M_2 exists at which the values of cooling effectiveness of the two types are equal. This value of M_1/M_2 is apparently a function of $\sigma_{1_{av}} \Delta p_1$, $\sigma_{2_{av}} \Delta p_2$, and m.

The factor $Nd l_2/M_2$ is an index of tube weight and also, for a given tube diameter, an index of volume taken up by the tubes $\left(\frac{\pi d}{4} \frac{Nd l_2}{M_2}\right)$. For a given tube diameter and for a given $Nd l_2/M_2$, therefore, the difference in volume between the two types of intercoolers is the difference in the volume of the spacing between the tubes. For a given $\frac{s_p}{s + d_o}$ (that is, a given staggered tube arrangement) and within the range of available test data, the relation between the tube spacings s_a and s_t for a charge-across-tube type and charge-through-tube type, respectively, may be expressed as (see equation (15)),

$$\frac{s_t}{s_a} = K_5 \left[\frac{\left(\sigma_{2_{av}} \Delta p_2\right)_a}{\left(\sigma_{1_{av}} \Delta p_1\right)_t}\right]^{0.5} \left[\frac{(M_1)_t}{(M_2)_a}\right]^{0.89}$$

where K_s is approximately equal to 1.

In order to operate at a high cooling effectiveness, an intercooler must usually handle a greater weight flow of cooling air than charge, and the pressure drop allowed for the charge is usually greater than that available for the cooling air. It is then evident, from the foregoing equation, that the tube spacing and thus the volumetric requirements of the charge-through-tube type will usually be greater than those of the other type.

CONCLUDING REMARKS

With the aid of heat-transfer theory, relationships between intercooler performance and various intercooler dimensions may be derived for tubular intercoolers of the charge-through-tube type. The graphical representation of these relationships in the form of design charts included in this report simplifies the correlation of the many variables involved and should be of material assistance in the selection of an intercooler to satisfy a particular set of conditions.

With a given staggered arrangement of tubes and in the range of practical intercooler operation, the charge-through-tube type of intercooler generally requires more space than the charge-across-tube type.

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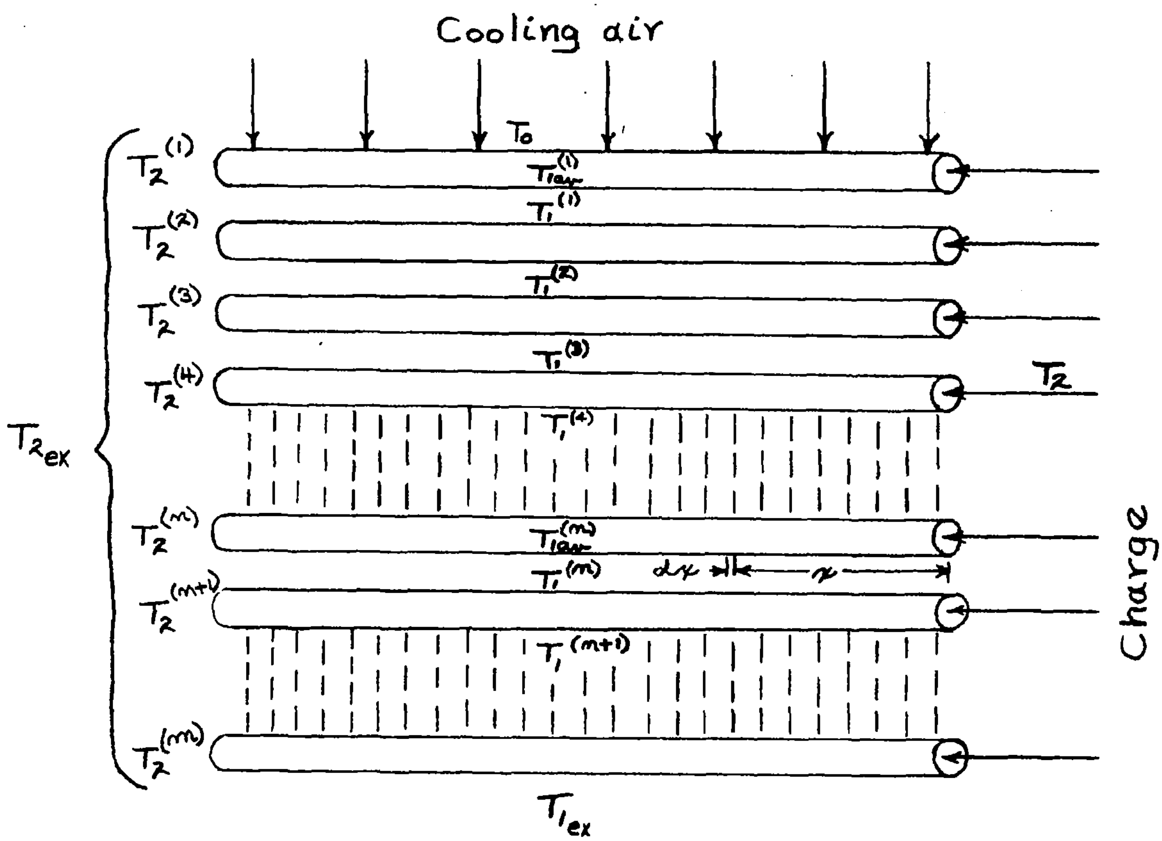


Figure 1.- Diagram of intercooler showing symbols used in the analysis.

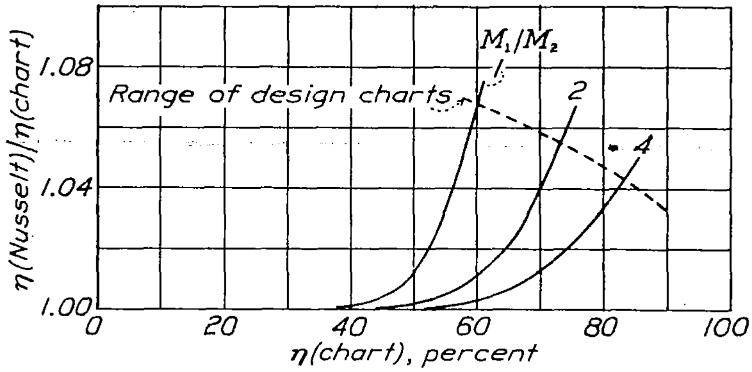
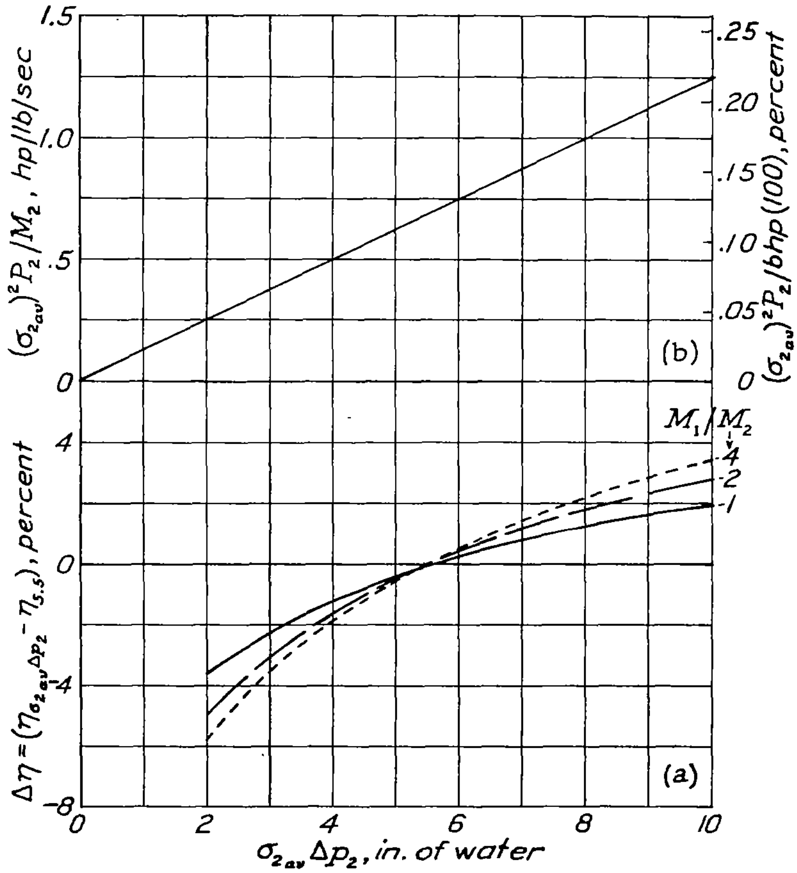


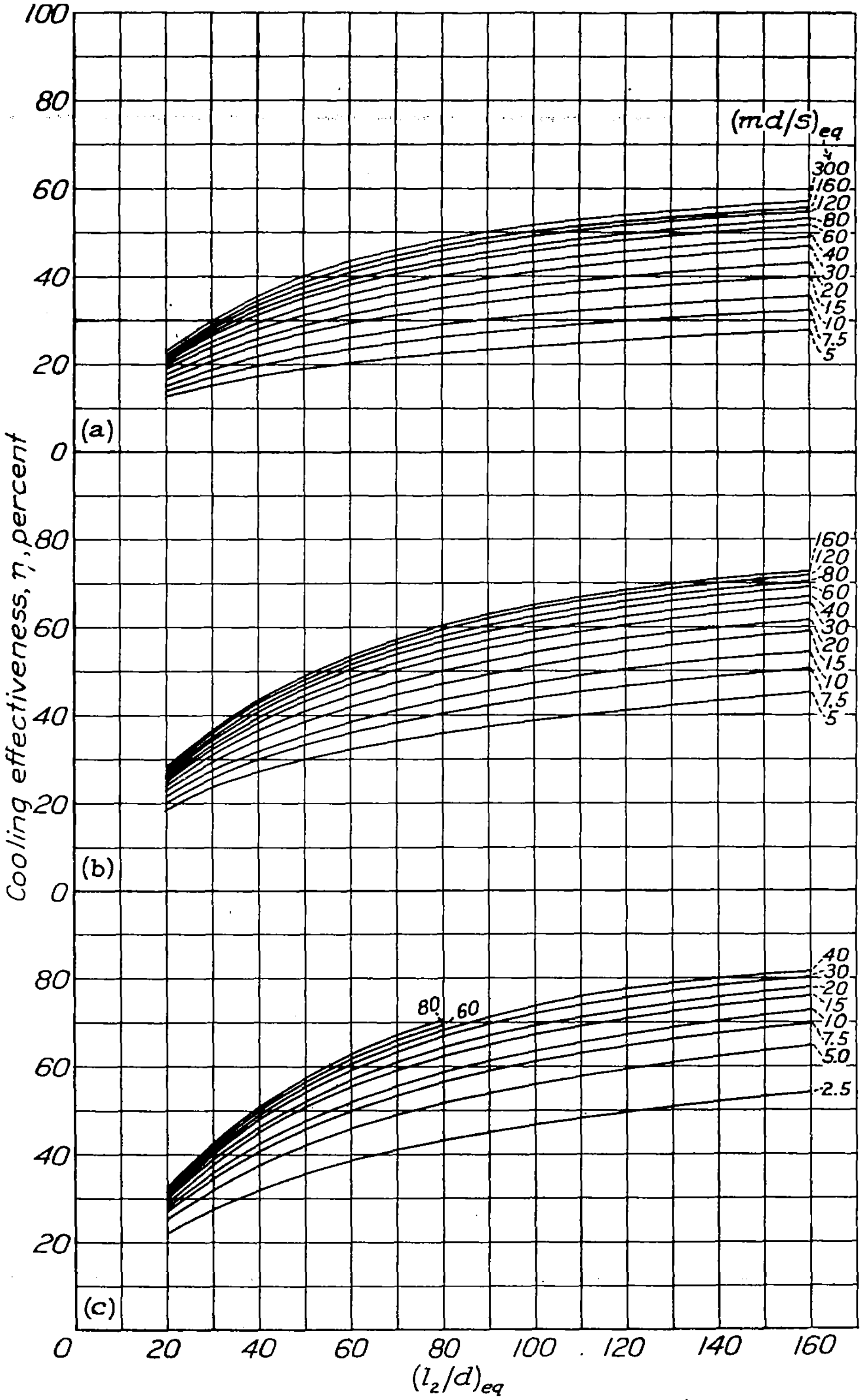
Figure 2.- A comparison of cooling effectiveness obtained by Nusselt with those obtained from the design charts.



(a) Cooling effectiveness correction variation with charge pressure drop and M_1/M_2.

(b) Variation in the power required to force the charge through the intercooler with charge pressure drop.

Figure 9.- Relation between charge pressure drop and intercooler performance.



(a) M_1/M_2,1; (b) M_1/M_2,2; (c) M_1/M_2,4
Figure 3.- Effect of intercooler dimensions on cooling effectiveness.

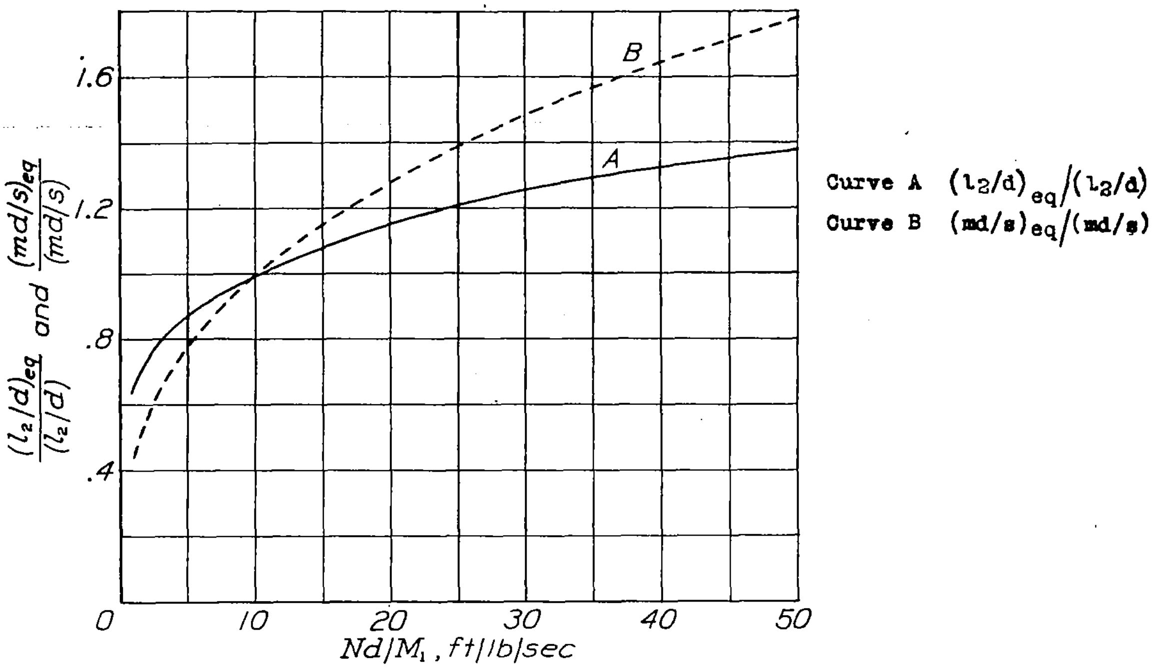


Figure 4.- Correction factors for l_2/d and md/s (from reference 1).

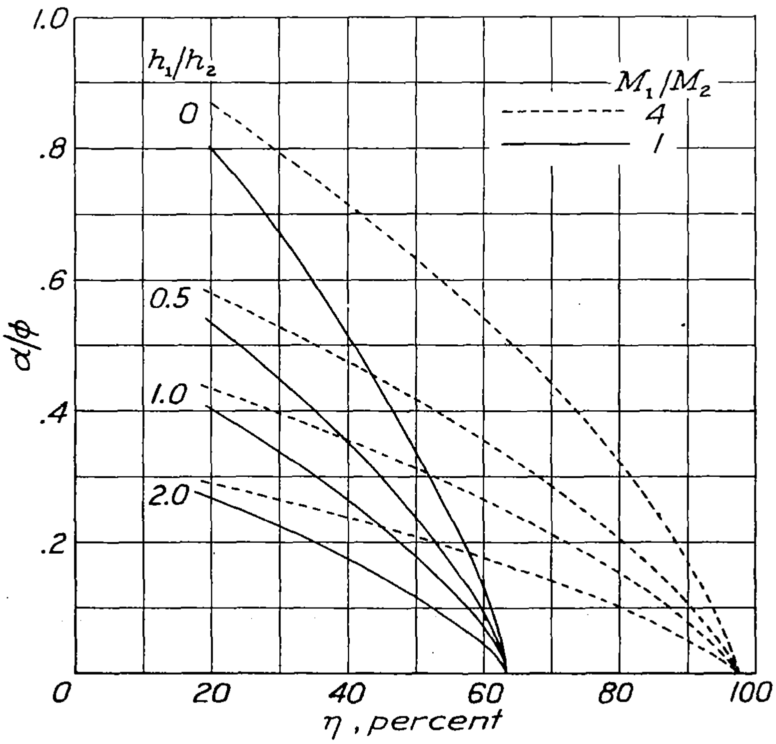
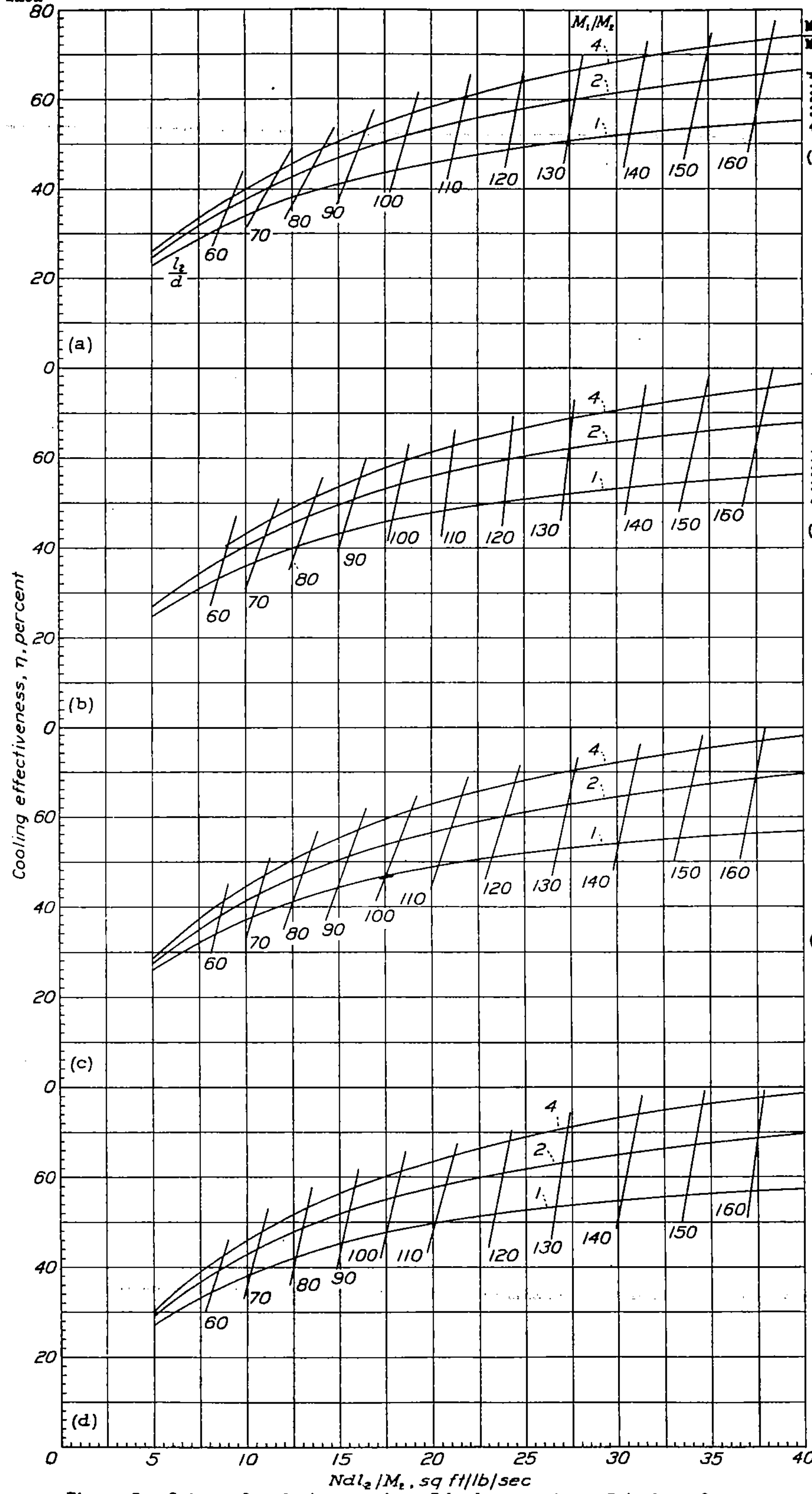


Figure 13.- Effect of surface heat-transfer coefficients on cooling effectiveness.



$\frac{M_1}{M_2}$	$(\sigma_{1_{av}})^2 \frac{P_1}{M_2}$ (hp/lb/sec)	$(\sigma_{1_{av}})^2 \frac{P_1(100)}{bhp}$ (percent)
1	0.24	0.043
2	.50	.086
3	.74	.128
4	.99	.171

(a) $\sigma_{1_{av}}\Delta p_1$,2 inches of water

1	0.50	0.086
2	.99	.171
3	1.48	.257
4	1.97	.342

(b) $\sigma_{1_{av}}\Delta p_1$,4 inches of water

1	0.74	0.128
2	1.48	.257
3	2.23	.386
4	2.97	.514

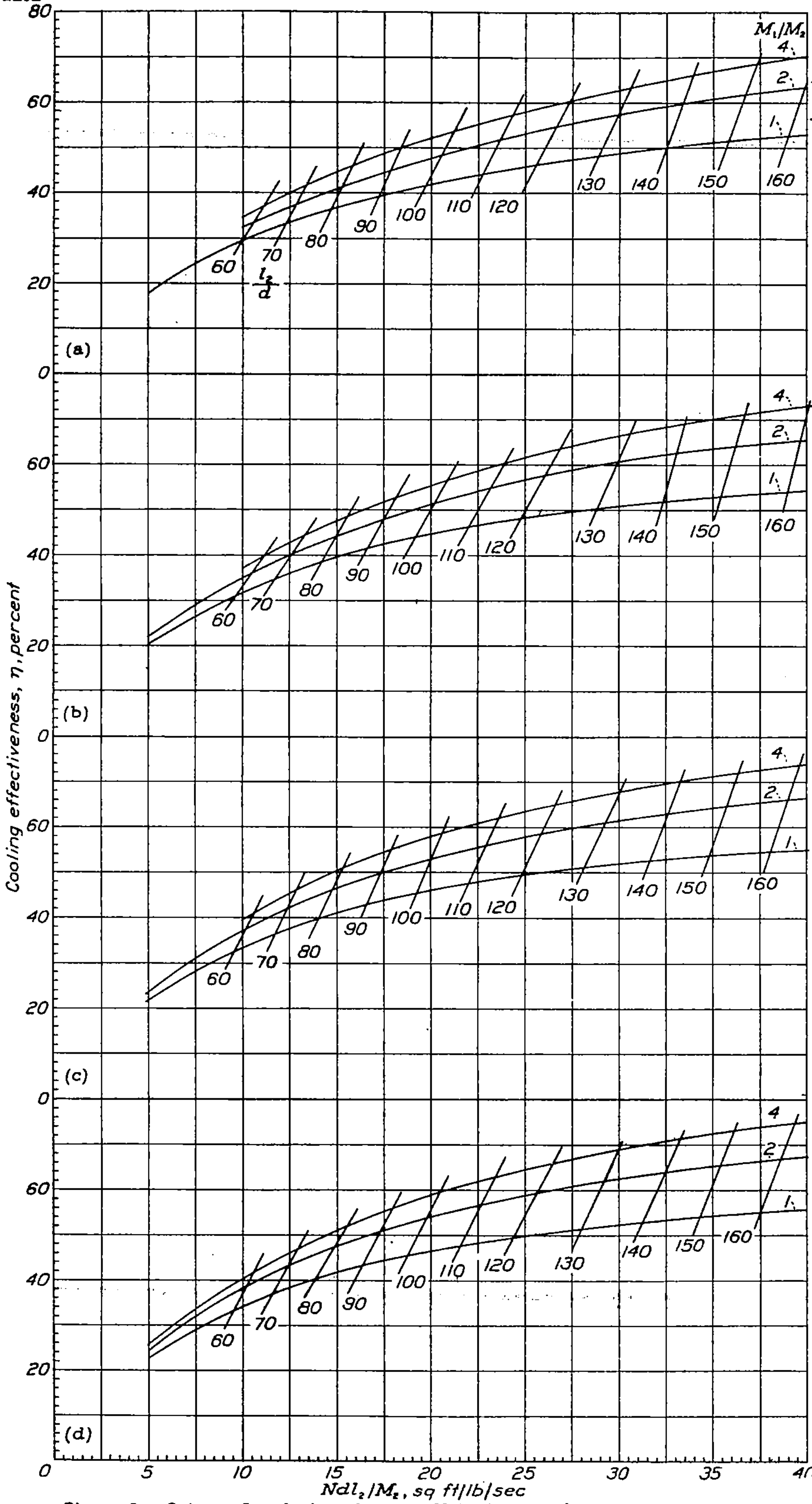
(c) $\sigma_{1_{av}}\Delta p_1$,6 inches of water

1	0.99	0.171
2	1.97	.342
3	2.97	.514
4	3.95	.685

$(\sigma_{2_{av}})^2 \frac{P_2}{M_2}$	$(\sigma_{2_{av}})^2 \frac{P_2(100)}{bhp}$
0.68	0.118

(d) $\sigma_{1_{av}}\Delta p_1$,8 inches of water

Figure 5.- Intercooler design chart. m,5 banks; $\sigma_{2_{av}}\Delta p_2$,5.5 inches of water.



$\frac{M_1}{M_2}$	$(\sigma_{1_{av}})^2 \frac{P_1}{M_2}$ (hp/lb/sec)	$(\sigma_{1_{av}})^2 \frac{P_1(100)}{bhp}$ (percent)
1	0.24	0.043
2	.50	.086
3	.74	.128
4	.99	.171

(a) $\sigma_{1_{av}}\Delta p_1$, 2 inches of water

1	0.50	0.086
2	.99	.171
3	1.48	.257
4	1.97	.342

(b) $\sigma_{1_{av}}\Delta p_1$, 4 inches of water

1	0.74	0.128
2	1.48	.257
3	2.23	.386
4	2.97	.514

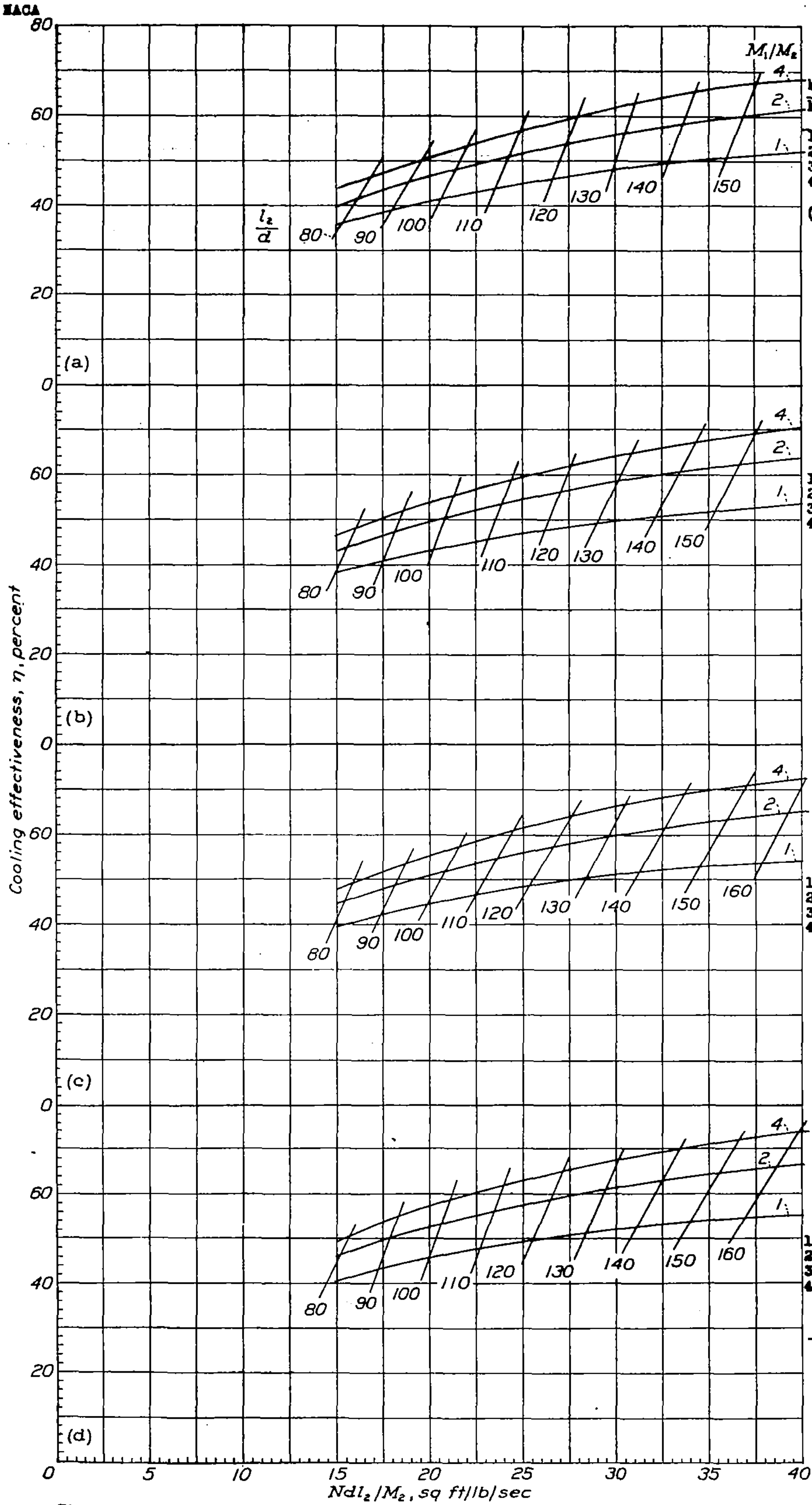
(c) $\sigma_{1_{av}}\Delta p_1$, 6 in of water

1	0.99	0.171
2	1.97	.342
3	2.97	.514
4	3.95	.685

$(\sigma_{2_{av}})^2 \frac{P_2}{M_2}$	$(\sigma_{2_{av}})^2 \frac{P_2(100)}{bhp}$
0.68	0.118

(d) $\sigma_{1_{av}}\Delta p_1$, 8 inches of water

Figure 8.- Intercooler design chart. m,20 banks; $\sigma_{2_{av}}\Delta p_2$, 5.5 inches of water



$\frac{M_1}{M_2}$	$(\sigma_{1_{av}})^2 \frac{P_1}{M_2}$ (hp/lb/sec)	$(\sigma_{1_{av}})^2 \frac{P_1(100)}{bhp}$ (percent)
1	0.24	0.043
2	.50	.086
3	.74	.128
4	.99	.171

(a) $\sigma_{1_{av}} \Delta p_1$, 2 inches of water

1	0.50	0.086
2	.99	.171
3	1.48	.257
4	1.97	.342

(b) $\sigma_{1_{av}} \Delta p_1$, 4 inches of water

1	0.74	0.128
2	1.48	.257
3	2.23	.386
4	2.97	.514

(c) $\sigma_{1_{av}} \Delta p_1$, 6 inches of water

1	0.99	0.171
2	1.97	.342
3	2.97	.514
4	3.95	.685

$(\sigma_{2_{av}})^2 \frac{P_2}{M_2}$	$(\sigma_{2_{av}})^2 \frac{P_2(100)}{bhp}$
0.68	0.118

(d) $\sigma_{1_{av}} \Delta p_1$, 8 inches of water

Figure 7.- Intercooler design chart. m,30 banks; $\sigma_{2_{av}} \Delta p_2$, 5.5 inches of water.

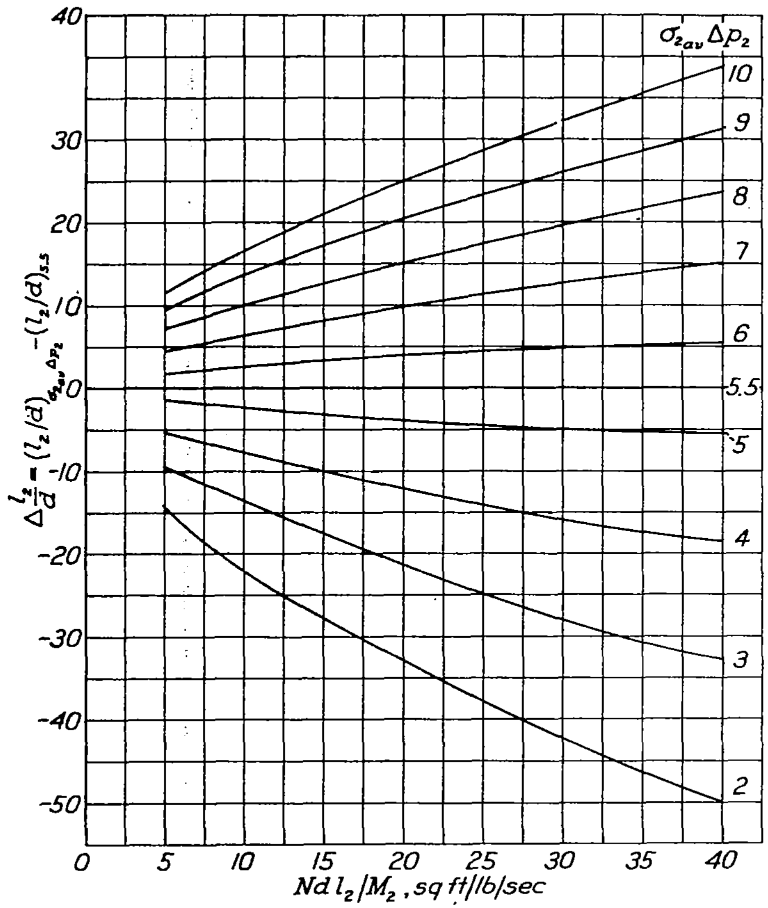
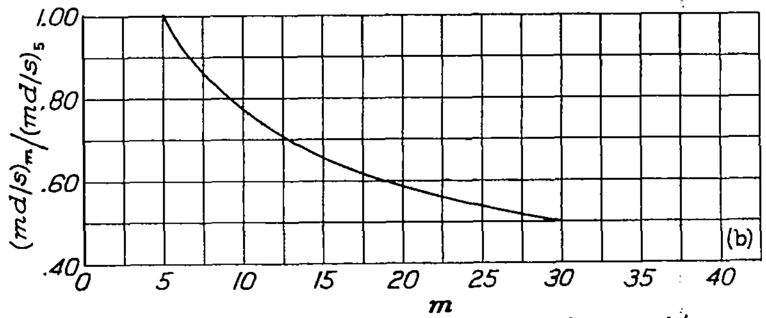
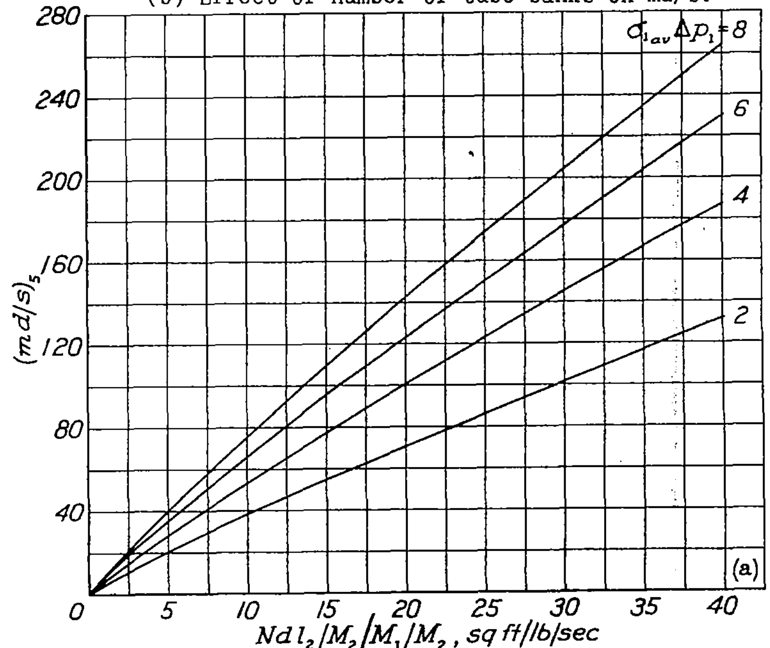


Figure 8.- Relation between charge pressure drop, Ndl_2/M_2, and intercooler tube length-diameter ratio.

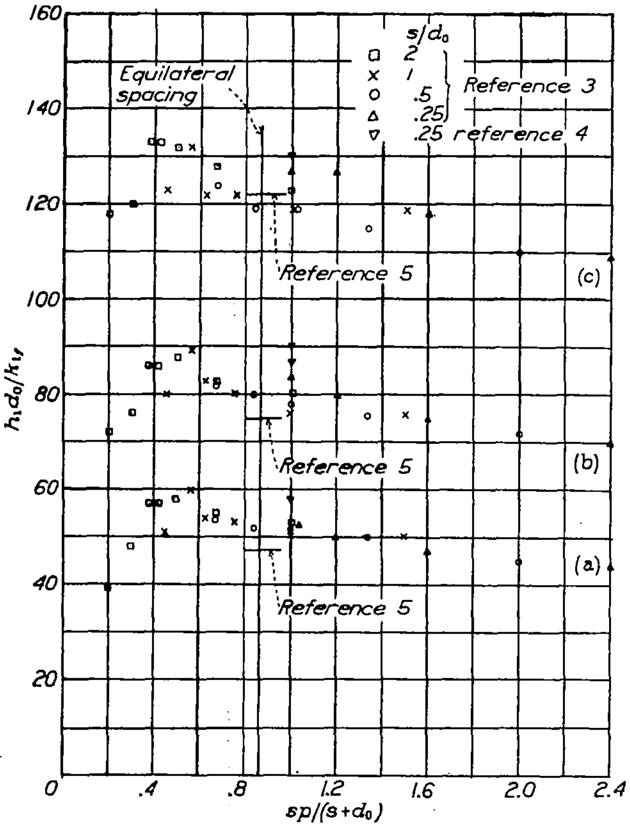


(b) Effect of number of tube banks on md/s.



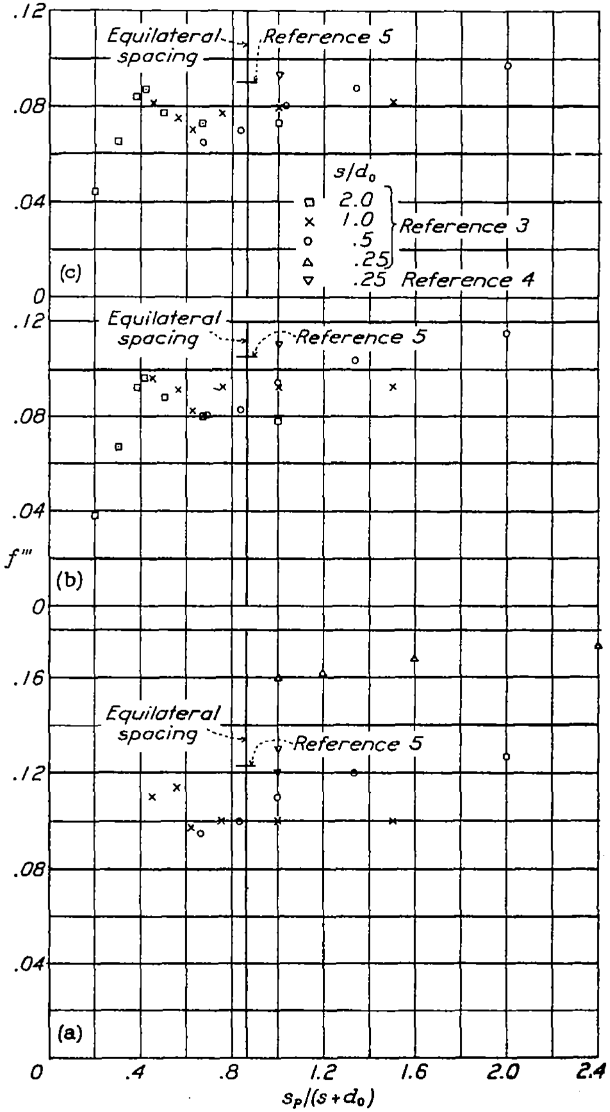
(a) Relation between the cooling-air pressure drop, Ndl_2/M_2, M_1/M_2, and md/s for m = 5 banks.

Figure 10.- Effect of cooling-air pressure drop, Ndl_2/M_2, and M_1/M_2 on md/s.



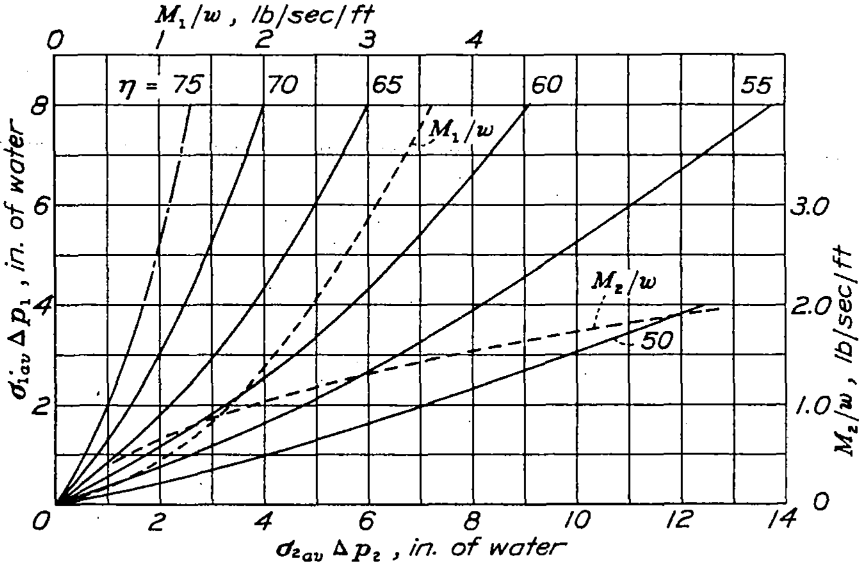
$$\rho l_{av} V l_{max} d_0/\mu l_f = \begin{array}{l}(a)\ \ 5{,}000\\ (b)\ 10{,}000\\ (c)\ 20{,}000\end{array}$$

Figure 11.- Effect of tube arrangement on Nusselt number in flow across staggered tube banks (from references 3 and 4).



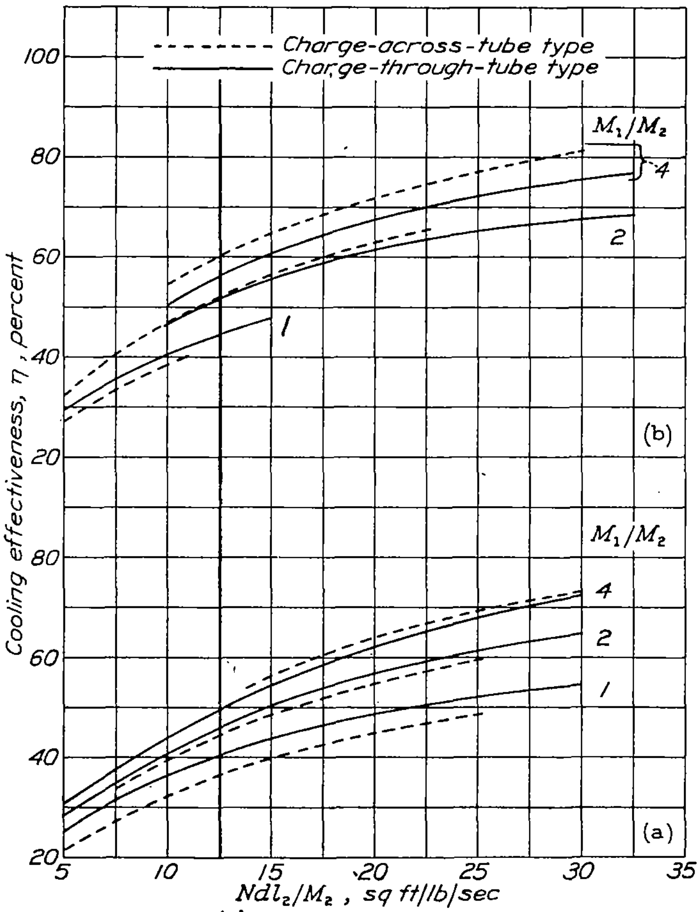
$$\rho l_{av} V l_{max} s/\mu l_f = \begin{array}{l}(a)\ \ 5{,}000\\ (b)\ 10{,}000\\ (c)\ 20{,}000\end{array}$$

Figure 12.- Effect of tube arrangement on friction factor in flow across staggered tube banks (from references 3 and 4).



d = .0208 ft
m = 20 banks
l_2 = 2.65 ft
s = .00958 ft
N = 647w

Figure 14.- Sample performance chart of an intercooler (case 1).



(a) $\sigma_{1_{av}}\Delta p_1$, 2 inches of water

(b) $\sigma_{1_{av}}\Delta p_1$, 8 inches of water

Figure 15.- Comparison of two types of cross-flow tubular intercoolers. m, 5 banks, $\sigma_{2_{av}}\Delta p_2$, 10 inches of water.